Charles Darwin University

Final Examination

Family Name	
Given Name/s	
Student Number	
Teaching Period	Semester 1, 2018

ENG487 – HVAC Systems	DURATION	
	Reading Time:	10 minutes
	Writing Time:	180 minutes

INSTRUCTIONS TO CANDIDATES

This examination consists of 6 (six) questions. Candidates are required to answer **ALL**.

The total mark for this examination is **40 marks**.

EXAM CONDITIONS

You may begin writing from the commencement of the examination session. The reading time indicated above is provided as a guide only.

This is a CLOSED BOOK examination

Any non-programmable calculator is permitted

No handwritten notes are permitted

No dictionaries are permitted

ADDITIONAL AUTHORISED MATERIALS	EXAMINATION MATERIALS TO BE SUPPLIED
No additional printed material is permitted	1 x 20 Page Book Formula Sheet/s

THIS EXAMINATION IS PRINTED DOUBLE-SIDED.

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Question 1:

A classroom is designed for 150 people. Assume that each person breaths out CO_2 at the rate of 0.35 L/min. The concentration of CO_2 in the incoming air is 280 ppm (0.028 percent). It is required to hold the concentration in the room below 1000 ppm (0.1 percent).

Assuming the air in the room is perfectly mixed, what is the minimum rate of air flow required to maintain the desired level? (4 marks)

Question 2:

A dancing hall has occupancy of 45 people engaged in moderate dancing activities from 8 am to 5 pm. The average light level is 15 W/m^2 of vented fluorescent fixtures with a ceiling plenum return. Electronic equipment amounts to 7 kW. Estimate the sensible and latent heat gain to the space for a floor area of 750 m^2 at 4 pm. For the sensible hear gain, estimate the radiative and convective portions.

Assume:

- For the sensible heat gain, assume 70% radiative to 30% convective for people and 20% radiative to 80% convective for equipment.
- 50% of lighting to enter plenum space directly, with the remaining to be 59% radiative to 41% convective. (10 marks)

Question 3:

What is the best thermostat setting (air dry bulb temperature) in a factory where the workmen are standing, walking, lifting and performing various machine tasks (met = 2)? Calculate the answer in F or ^{o}C.

Assume:

- A globe temperature measurement reads 72 F (22 ^{o}C), the relative humidity will be in the 45% range, and air motion will likely be around 0.15 m/s.
- The men are dressed in typical summer garments (clo = 0.5). (10 marks)

Question 4:

Air enters a cooling coil at the rate of 3 m^3/s at 30 ^{o}C dry bulb, 22 ^{o}C wet bulb and sea-level pressure. The air leaves the coil at 13 ^{o}C dry bulb, 12 ^{o}C wet bulb.

(a) Show all the conditions on the psychrometric chart (provided in page 5-8). (2 marks)

(b) Determine the SHF and the apparatus dew point. (2 marks)

(c) Compute the total and sensible heat transfer rates from the air. (2 marks)

Question 5:

A wall is 20 ft wide and 8 ft high and has an overall heat-transfer coefficient of 0.4 W/m^2C. It contains a solid urethane foam core steel door with thermal break, 80 x 32 x 1.75 in., and a double glass window (6.4 mm air space), 120 x 30 in. The window is metal sash with thermal break. Assuming parallel heat-flow paths for the wall, door and window, find the overall thermal resistance and overall heat-transfer coefficient for the combination. Assume winter conditions.

(5 marks)

Question 6: (5 marks)

A centrifugal fan is delivering 0.94 m^3/s of air at a total pressure differential of 473 Pa. The fan has an outlet area of 0.08 m^2 and requires 0.8 kW shaft input. Compute:

(a) the total power (1 marks)

(b) the total efficiency (1 marks)

(c) the fan static pressure (2 marks)

(d) the static efficiency (1 marks)

Chart 1*b* ASHRAE Psychrometric Chart No. 1 (SI) (Reprinted by permission of ASHRAE.)

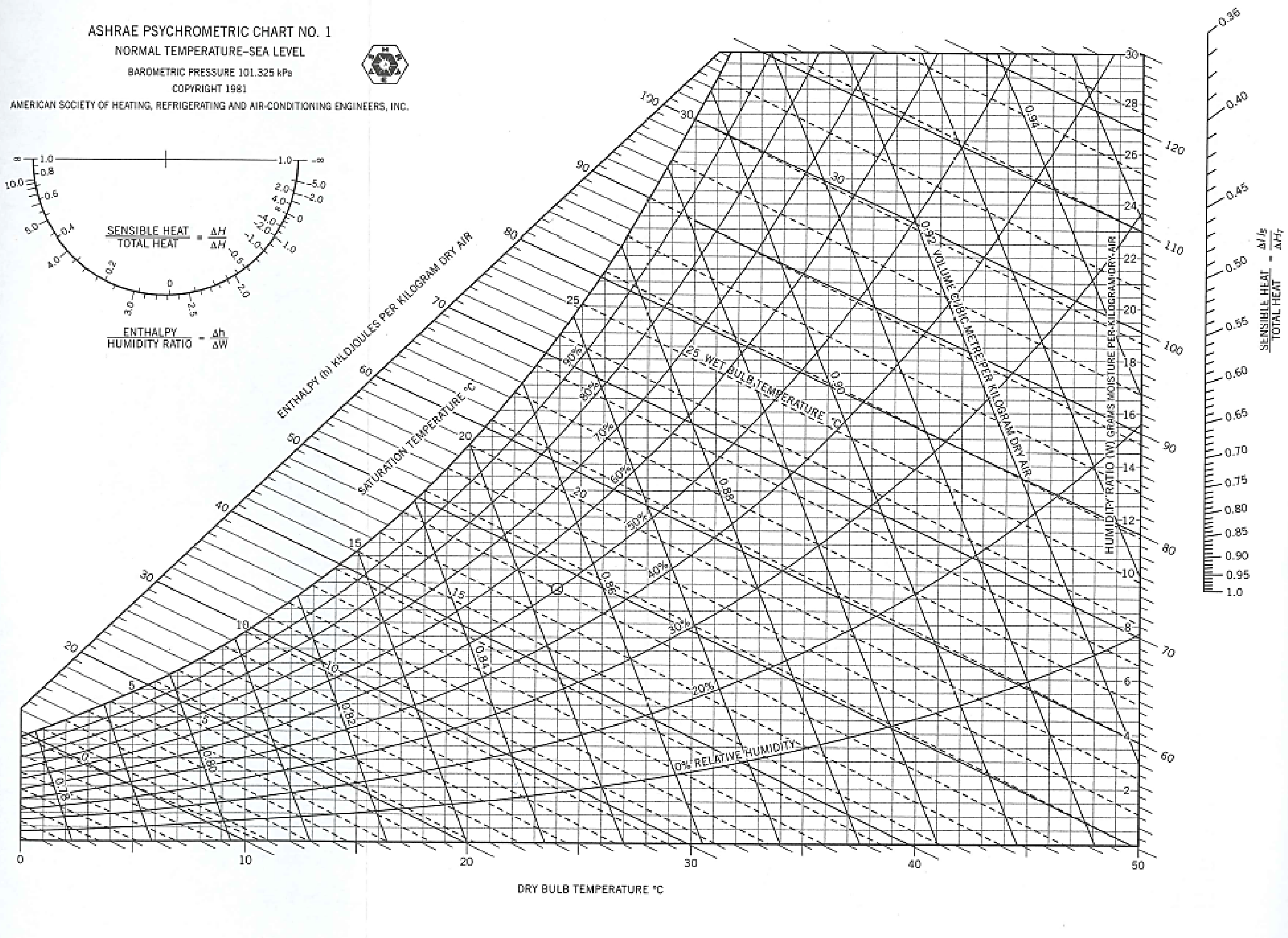


Chart 1*b* ASHRAE Psychrometric Chart No. 1 (SI) (Reprinted by permission of ASHRAE.)

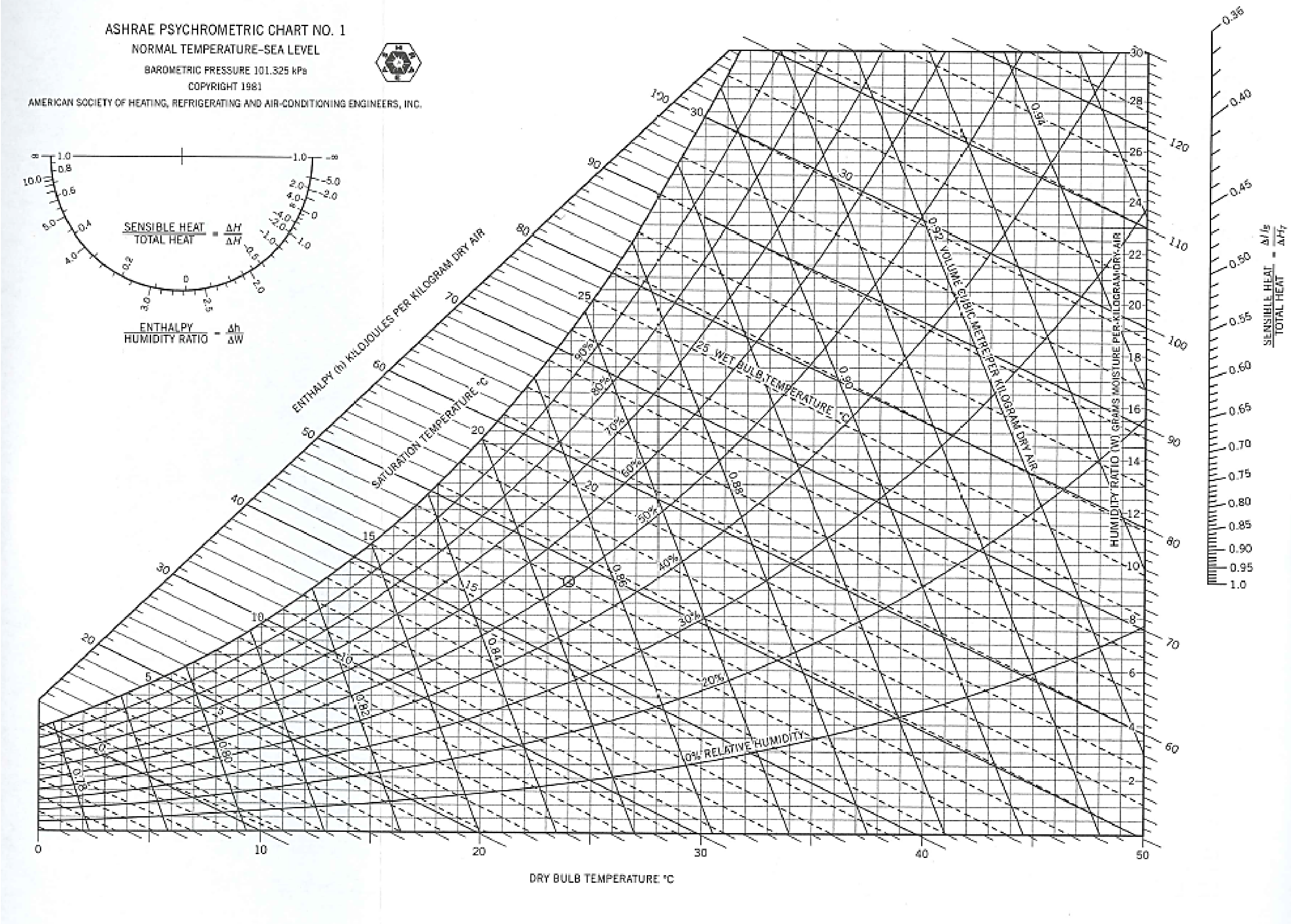


Chart 1*b* ASHRAE Psychrometric Chart No. 1 (SI) (Reprinted by permission of ASHRAE.)

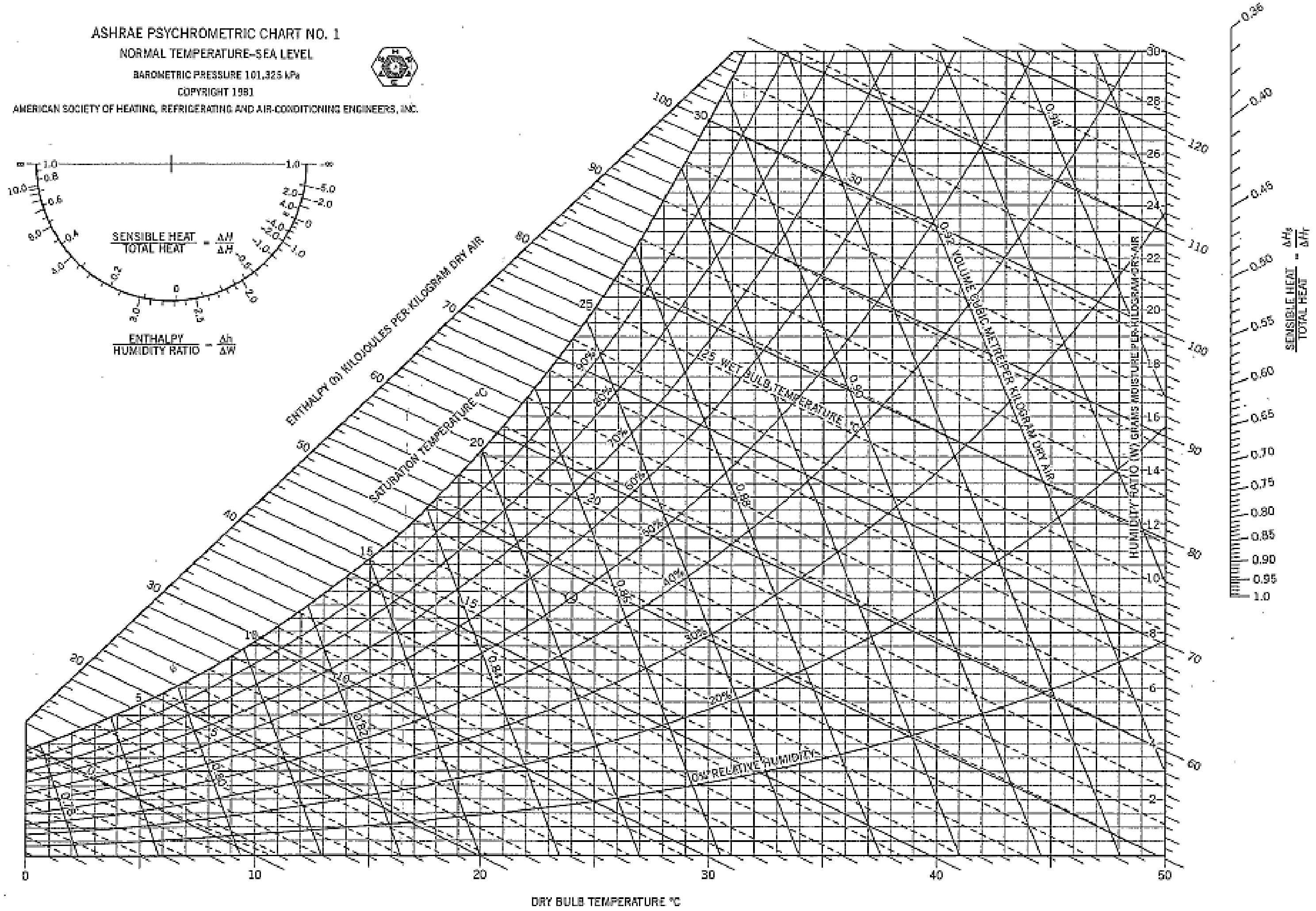
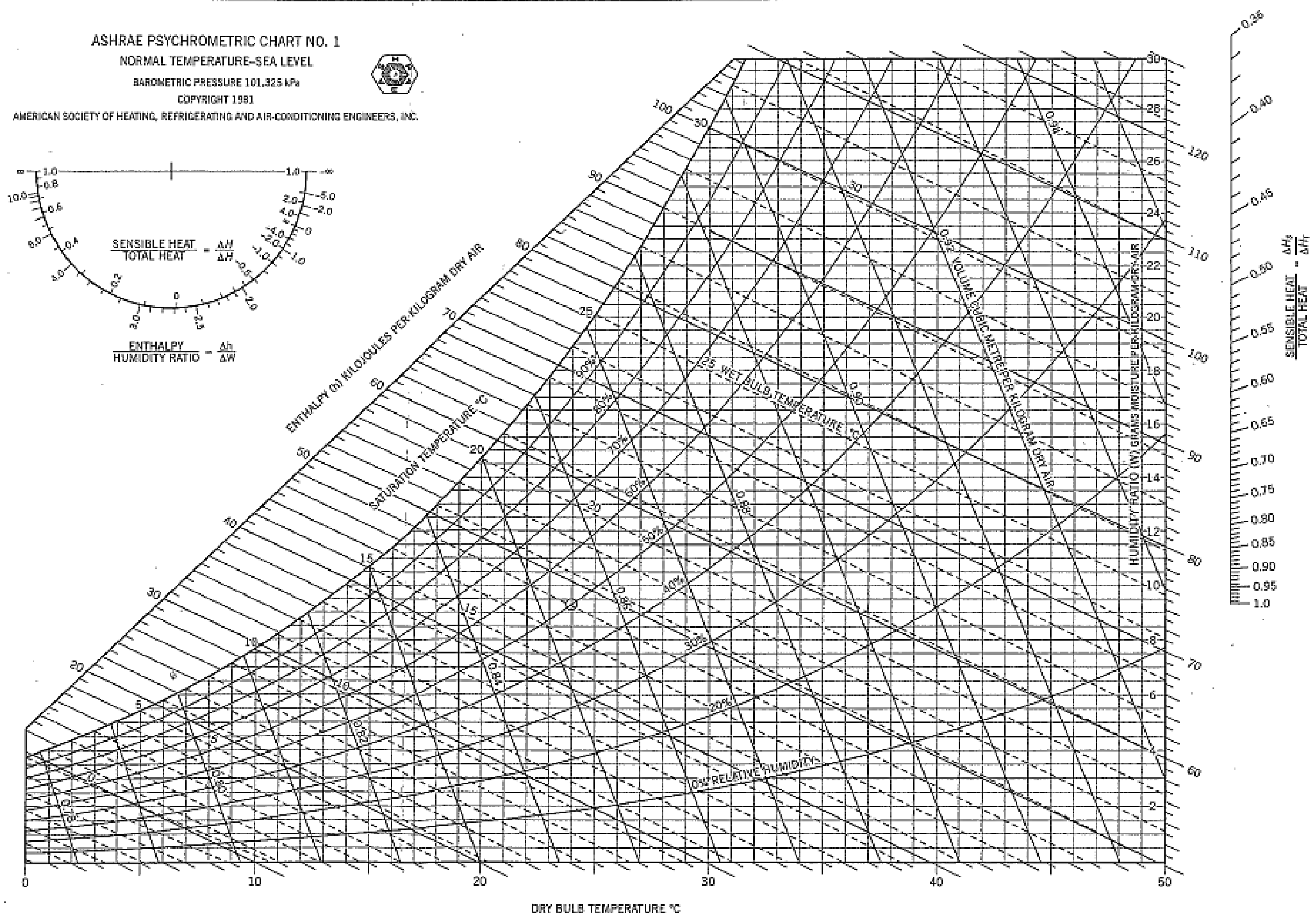


Chart 1*b* ASHRAE Psychrometric Chart No. 1 (SI) (Reprinted by permission of ASHRAE.)



Chapter 4 equation:

$$T_{mrt}^4 = T_g^4 + C\bar{V}^{1/2}(T_g - T_a) \quad (4\text{-}1)$$

where

T_{mrt} = mean radiant temperature, R or K
T_g = globe temperature, R or K
T_a = ambient air temperature, R or K
$\bar{V}$ = air velocity, fpm or m/s
$C = 0.103 \times 10^9$ (English units) $= 0.247 \times 10^9$ (SI units)

0 °C=273.15 K

Acceptable operative temperatures for active persons can be calculated (for 1.2 < met < 3) in degrees Fahrenheit from:

$$t_{o,active} = t_{o,sedentary} - 5.4\,(1 + \text{clo})(\text{met} - 1.2)\ \text{F} \quad (4\text{-}4a)$$

In degrees Celsius from:

$$t_{o,active} = t_{o,sedentary} - 3.0\,(1 + \text{clo})(\text{met} - 1.2)\ \text{C} \quad (4\text{-}4b)$$

$$\dot{Q}_t C_e + \dot{N} = \dot{Q}_t C_s \quad (4\text{-}5)$$

where:

$\dot{Q}_t$ = rate at which air enters or leaves the space
C_s = average concentration of a contaminant within the space
$\dot{N}$ = rate of contaminant generation within the space
C_e = concentration of the contaminant of interest in the entering air

Equation 4-5 can be solved for the concentration level in the space C_s or for the necessary rate $\dot{Q}_t$ at which air must enter the space to maintain the desired concentration level of a contaminant within the space. This fundamental equation may be used as the basis for deriving more complex equations for more realistic cases.

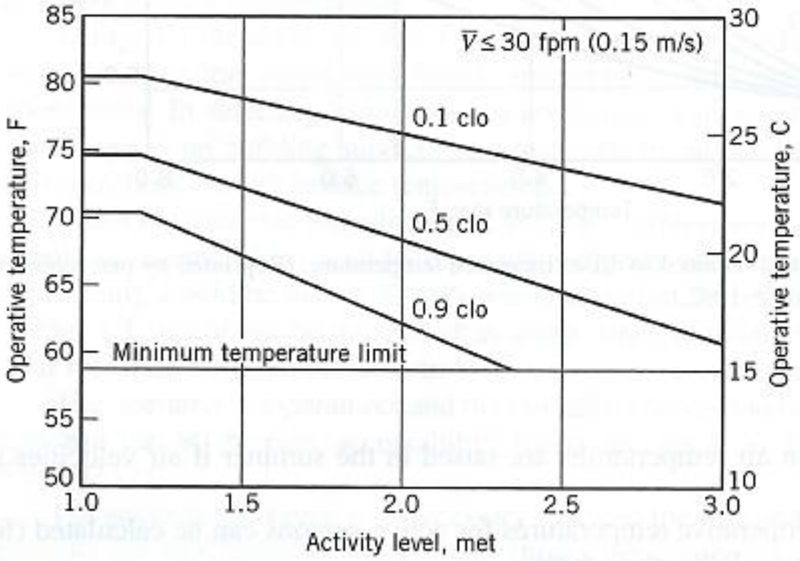


Figure 4-4 Optimum operative temperatures for active people in low-air-movement environments ($\overline{V} < 30$ fpm or 0.15 m/s). (Reprinted by permission from ASHRAE Standard 55-1992.)

Chapter 12 equation:

of Eq. 12-1b by the mass flow rate of the air produces an expression for the *total power* imparted to the air:

$$\dot{W}_t = \dot{m}\frac{\left(P_{01} - P_{02}\right)}{\rho} \tag{12-2}$$

The *static power* is the part of the total power that is used to produce the change in static pressure:

$$\dot{W}_s = \frac{\dot{m}\left(P_1 - P_2\right)}{\rho} = \dot{Q}\left(P_1 - P_2\right) \tag{12-3}$$

where $\dot{Q}$ = volume flow rate, ft^3/min or m^3/s.

Fan efficiency may be expressed in two ways. The total fan efficiency is the ratio of total air power $\dot{W}_t$ to the shaft power input $\dot{W}_{sh}$:

$$\eta_t = \frac{\dot{W}_t}{\dot{W}_{sh}} = \frac{\dot{m}\left(P_{01} - P_{02}\right)}{\rho \dot{W}_{sh}} = \dot{Q}\,\frac{P_{01} - P_{02}}{\dot{W}_{sh}} \tag{12-4a}$$

It has been common practice in the United States for $\dot{Q}$ to be in ft^3/min, $P_{01} - P_{02}$ to be in in. wg, and $\dot{W}_{sh}$ to be in horsepower. In this special case

$$\eta_t = \frac{\dot{Q}\left(P_{01} - P_{02}\right)}{6350\dot{W}_{sh}} \tag{12-4b}$$

The *static fan efficiency* is the ratio of the static air power to the shaft power input:

$$\eta_s = \frac{\dot{W}_s}{\dot{W}_{sh}} = \frac{\dot{m}\left(P_1 - P_2\right)}{\rho \dot{W}_{sh}} = \frac{\dot{Q}\left(P_1 - P_2\right)}{\dot{W}_{sh}} \tag{12-5a}$$

Using the units of Eq. 12-4b, we get

$$\eta_s = \frac{\dot{Q}\left(P_1 - P_2\right)}{6350\dot{W}_{sh}} \tag{12-5b}$$

In this form each term has the units of pressure in any system of units. For air at standard conditions and English units, pressure is usually in in. wg:

$$P_v = \rho\left(\frac{\overline{V}}{1097}\right)^2 = \left(\frac{\overline{V}}{4005}\right)^2 \text{ in. wg} \tag{12-10}$$

where $\overline{V}$ is in ft/min and ρ is in lbm/ft^3. In SI units,

$$P_v = \rho\left(\frac{\overline{V}}{1.414}\right)^2 = \left(\frac{\overline{V}}{1.29}\right)^2 \text{ Pa} \tag{12-11}$$

where $\overline{V}$ is in m/s and ρ is in kg/m^3. The mass density ρ is assumed equal to 62.4 lbm/ft^3 and 999 kg/m^3, respectively, in the last terms of Eqs. 12-10 and 12-11.

Table D-1 Conversion Factors

Length
1 ft = 30.48 cm
1 in. = 2.54 cm
1 m = 39.37 in.
1 micron = 10^{-6} m = 3.281 x 10^{-6} ft
1 mile = 5280 ft
Area
1 m^2 = 1550.1472 $in.^2$
1 m^2 = 10.76392 ft^2
Volume
1 ft^3 = 7.48 U.S. gallons = 1728 $in.^3$
1 m^3 = 6.1 x 10^4 $in.^3$
1 m^3 = 35.3147 ft^3
1 m^3 = 264.154 U.S. gallons
Mass
1 kg = 2.20462 lbm
1 lbm = 7000 grains = 453.5924 g
Force
1 N = 0.224809 lbf
1 lbf = 4.44822 N
Energy
1 Btu = 778.28 ft-lbf
1 Kilocalorie = 10^3 calories = 3.968 Btu
1 J = 9.48 x 10^{-4} Btu = 0.73756 ft-lbf
1 kW-hr = 3412 Btu = 2.6552 x 10^6 ft-lbf
Power
1 hp = 33.000 (ft-lbf)/min
1 hp = 745.7 W
1 W = 3.412 Btu/hr = 0.001341 hp = 0.0002843 tons of refrigeration
Pressure
1 atm = 14.6959 psia = 2116 lbf/ft^2 = 101325 N/m^2
1 in. wg = 249.08 Pa
1 in. of mercury = 3376.85 Pa
1 lbf/$in.^2$ = 6894.76 Pa
1 Pa = 1 N/m^2 = 1.4504 x 10^{-4} lbf/$in.^2$

continues

Table D-1 Conversion Factors *(continued)*

Temperature

1 degree R difference = 1 degree F difference = $\frac{5}{9}$ degree C difference

= $\frac{5}{9}$ degree K difference

degrees F = $\frac{9}{5}$ (degrees C) + 32

0 °C = 273.15 K

degrees C = $\frac{5}{9}$ (degrees F − 32)

Thermal Conductivity

$1 \frac{\text{Btu}}{\text{hr-ft-F}} = 0.004134 \frac{\text{calorie}}{\text{s-cm-C}} = 1.7307 \frac{\text{W}}{\text{m-C}}$

$1 \frac{\text{W}}{\text{m-C}} = 0.5778 \frac{\text{Btu}}{\text{hr-ft-F}}$

$1 \frac{\text{Btu-in}}{\text{hr-ft}^2\text{-F}} = 0.1442 \frac{\text{W}}{\text{m-C}}$

$1 \frac{\text{W}}{\text{m-C}} = 6.933 \frac{\text{Btu-in}}{\text{hr-ft}^2\text{-F}}$

Heat Transfer Coefficient

$1 \frac{\text{Btu}}{\text{hr-ft}^2\text{-F}} = 5.678 \frac{\text{W}}{\text{m}^2\text{-C}}$

$1 \frac{\text{W}}{\text{m}^2\text{-C}} = 0.1761 \frac{\text{Btu}}{\text{hr-ft}^2\text{-F}}$

Viscosity Absolute

1 poise = 100 centipoises

$1 \frac{\text{lbm}}{\text{sec - ft}}$ = 1490 centipoises = $1.49 \frac{\text{N - s}}{\text{m}^2}$

$1 \frac{\text{lbf - sec}}{\text{ft}^2}$ = 47,800 centipoises

1 centipoise = $0.001 \frac{\text{N - s}}{\text{m}^2}$

Viscosity Kinematic

1 ft^2/sec = 0.0929 m^2/s

1 m^2/s = 10.764 ft^2/sec

Specific Heat

$1 \frac{\text{calorie}}{\text{g - C}} = 1 \frac{\text{Btu}}{\text{lbm - F}}$

$1 \frac{\text{Btu}}{\text{lbm - F}} = 4186.8 \frac{\text{J}}{\text{kg - C}}$

$1 \frac{\text{J}}{\text{kg - C}} = 0.2388 \frac{\text{Btu}}{\text{lbm - F}}$

Table B.4

Physical Properties of Air at Standard Atmospheric Pressure (SI Units)[a]

Temperature (°C)	Density, ρ (kg/m^3)	Specific Weight[b], γ (N/m^3)	Dynamic Viscosity, μ (N·s/m^2)	Kinematic Viscosity, ν (m^2/s)	Specific Heat Ratio, k (—)	Speed of Sound, c (m/s)
−40	1.514	14.85	1.57 E − 5	1.04 E − 5	1.401	306.2
−20	1.395	13.68	1.63 E − 5	1.17 E − 5	1.401	319.1
0	1.292	12.67	1.71 E − 5	1.32 E − 5	1.401	331.4
5	1.269	12.45	1.73 E − 5	1.36 E − 5	1.401	334.4
10	1.247	12.23	1.76 E − 5	1.41 E − 5	1.401	337.4
15	1.225	12.01	1.80 E − 5	1.47 E − 5	1.401	340.4
20	1.204	11.81	1.82 E − 5	1.51 E − 5	1.401	343.3
25	1.184	11.61	1.85 E − 5	1.56 E − 5	1.401	346.3
30	1.165	11.43	1.86 E − 5	1.60 E − 5	1.400	349.1
40	1.127	11.05	1.87 E − 5	1.66 E − 5	1.400	354.7
50	1.109	10.88	1.95 E − 5	1.76 E − 5	1.400	360.3
60	1.060	10.40	1.97 E − 5	1.86 E − 5	1.399	365.7
70	1.029	10.09	2.03 E − 5	1.97 E − 5	1.399	371.2
80	0.9996	9.803	2.07 E − 5	2.07 E − 5	1.399	376.6
90	0.9721	9.533	2.14 E − 5	2.20 E − 5	1.398	381.7
100	0.9461	9.278	2.17 E − 5	2.29 E − 5	1.397	386.9
200	0.7461	7.317	2.53 E − 5	3.39 E − 5	1.390	434.5
300	0.6159	6.040	2.98 E − 5	4.84 E − 5	1.379	476.3
400	0.5243	5.142	3.32 E − 5	6.34 E − 5	1.368	514.1
500	0.4565	4.477	3.64 E − 5	7.97 E − 5	1.357	548.8
1000	0.2772	2.719	5.04 E − 5	1.82 E − 4	1.321	694.8

[a]Based on data from R. D. Blevins, *Applied Fluid Dynamics Handbook*, Van Nostrand Reinhold Co., Inc., New York, 1984.
[b]Density and specific weight are related through the equation $\gamma = \rho g$. For this table $g = 9.807$ m/s^2.

Table B.4: from Munson, Okiishi, Huebsch and Rothmayer, "Fluid Mechanics", 7th edition, Wiley

Table 5-5a *U*-Factors for Various Fenestration Products, Btu/(hr-ft^2-F) (Vertical Installation)[a]

	Frame: Glass Only		Operable (Including Sliding and Swinging Glass Doors)					Fixed
	Center of Glass	Edge of Glass	Aluminum without Thermal Break	Aluminum with Thermal Break	Reinforced Vinyl/ Aluminum-Clad Wood	Wood/ Vinyl	Insulated Fiberglass/ Vinyl	Insulated Fiberglass/ Vinyl
Single Glazing								
$\frac{1}{8}$ in. glass	1.04	1.04	1.27	1.08	0.90	0.89	0.81	0.94
$\frac{1}{4}$ in. acrylic/ polycarb	0.88	0.88	1.14	0.96	0.79	0.78	0.71	0.81
$\frac{1}{8}$ in. acrylic/ polycarb	0.96	0.96	1.21	1.02	0.85	0.83	0.76	0.87
Double Glazing								
$\frac{1}{4}$ in. air space	0.55	0.64	0.87	0.65	0.57	0.55	0.49	0.53
$\frac{1}{2}$ in. air space	0.48	0.59	0.81	0.60	0.53	0.51	0.44	0.48
$\frac{1}{4}$ in. argon space	0.51	0.61	0.84	0.62	0.55	0.53	0.46	0.50
Double Glazing, $\epsilon = 0.60$ on surface 2 or 3								
$\frac{1}{4}$ in. air space	0.52	0.62	0.84	0.63	0.55	0.53	0.47	0.51
$\frac{1}{2}$ in. air space	0.44	0.56	0.78	0.57	0.50	0.48	0.42	0.45
$\frac{1}{4}$ in. argon space	0.47	0.58	0.81	0.59	0.52	0.50	0.44	0.47
Double Glazing, $\epsilon = 0.10$ on surface 2 or 3								
$\frac{1}{4}$ in. air space	0.42	0.55	0.77	0.56	0.49	0.47	0.41	0.43
$\frac{1}{2}$ in. air space	0.32	0.48	0.69	0.49	0.42	0.40	0.35	0.35
$\frac{1}{4}$ in. argon space	0.35	0.50	0.71	0.51	0.44	0.42	0.36	0.37
$\frac{1}{2}$ in. argon space	0.27	0.44	0.65	0.45	0.39	0.37	0.31	0.31
Triple Glazing								
$\frac{1}{4}$ in. air space	0.38	0.52	0.72	0.51	0.44	0.43	0.38	0.40
$\frac{1}{2}$ in. air space	0.31	0.47	0.67	0.46	0.40	0.39	0.34	0.34
$\frac{1}{4}$ in. argon space	0.34	0.49	0.69	0.48	0.42	0.41	0.35	0.36
Triple Glazing, $\epsilon = 0.20$ on surfaces 2 or 3 and 4 or 5								
$\frac{1}{4}$ in. air space	0.29	0.45	0.65	0.44	0.38	0.37	0.32	0.32
$\frac{1}{2}$ in. air space	0.20	0.39	0.58	0.38	0.32	0.31	0.27	0.25
$\frac{1}{4}$ in. argon space	0.23	0.41	0.61	0.40	0.34	0.33	0.29	0.28
Triple Glazing, $\epsilon = 0.10$ on surfaces 2 or 3 and 4 or 5								
$\frac{1}{4}$ in. air space	0.27	0.44	0.64	0.43	0.37	0.36	0.31	0.31
$\frac{1}{2}$ in. air space	0.18	0.37	0.57	0.36	0.31	0.30	0.25	0.23
$\frac{1}{4}$ in. air space	0.21	0.39	0.59	0.39	0.33	0.32	0.27	0.26
Quadruple Glazing, $\epsilon = 0.10$ on surfaces 2 or 3 and 4 or 5								
$\frac{1}{4}$ in. air space	0.22	0.40	0.60	0.39	0.34	0.33	0.28	0.27

[a]Heat transmission coefficients are based on winter conditions of 0 F outdoors and 70 F indoors with 15 mph wind and zero solar flux. Small changes in the indoor and outdoor temperatures will not significantly affect the overall *U*-factors. Glazing layers are numbered from outdoor to indoor.

Source: Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.

Table 5-5b *U*-Factors for Various Fenestration Products, W/(m²-C) (Vertical Installation)[a]

	Frame: Glass Only: Center of Glass	Glass Only: Edge of Glass	Operable: Aluminum without Thermal Break	Operable: Aluminum with Thermal Break	Operable: Reinforced Vinyl/ Aluminum-Clad Wood	Operable: Wood/ Vinyl	Operable: Insulated Fiberglass/ Vinyl	Fixed: Insulated Fiberglass/ Vinyl
Single Glazing								
3.2 mm glass	5.91	5.91	7.24	6.12	5.14	5.05	4.61	5.35
6.4 mm acrylic/ polycarb	5.00	5.00	6.49	5.43	4.51	4.42	4.01	4.58
3.2 mm acrylic/ polycarb	5.45	5.45	6.87	5.77	4.82	4.73	4.31	4.97
Double Glazing								
6.4 mm air space	3.12	3.63	4.93	3.70	3.25	3.13	2.77	3.04
12.7 mm air space	2.73	3.36	4.62	3.42	3.00	2.87	2.53	2.72
6.4 mm argon space	2.90	3.48	4.75	3.54	3.11	2.98	2.63	2.85
Double Glazing, $\epsilon = 0.60$ on surface 2 or 3								
6.4 mm air space	2.95	3.52	4.80	3.58	3.14	3.02	2.67	2.90
12.7 mm air space	2.50	3.20	4.45	3.26	2.85	2.73	2.39	2.54
6.4 mm argon space	2.67	3.32	4.58	3.38	2.96	2.84	2.49	2.67
Double Glazing, $\epsilon = 0.10$ on surface 2 or 3								
6.4 mm air space	2.39	3.12	4.36	3.17	2.78	2.65	2.32	2.45
12.7 mm air space	1.82	2.71	3.92	2.77	2.41	2.28	1.96	1.99
6.4 mm argon space	1.99	2.83	4.05	2.89	2.52	2.39	2.07	2.13
12.7 mm argon space	1.59	2.49	3.70	2.56	2.22	2.10	1.79	1.76
Triple Glazing								
6.4 mm air space	2.16	2.96	4.11	2.89	2.51	2.45	2.16	2.25
12.7 mm air space	1.76	2.67	3.80	2.60	2.25	2.19	1.91	1.93
6.4 mm argon space	1.93	2.79	3.94	2.73	2.36	2.30	2.01	2.07
Triple Glazing, $\epsilon = 0.20$ on surfaces 2 or 3 and 4 or 5								
6.4 mm air space	1.65	2.58	3.71	2.52	2.17	2.12	1.84	1.84
12.7 mm air space	1.14	2.19	3.31	2.15	1.84	1.78	1.52	1.43
6.4 mm argon space	1.31	2.32	3.45	2.27	1.95	1.90	1.62	1.56
Triple Glazing, $\epsilon = 0.10$ on surfaces 2 or 3 and 4 or 5								
6.4 mm air space	1.53	2.49	3.63	2.44	2.10	2.05	1.77	1.75
12.7 mm air space	1.02	2.10	3.22	2.07	1.76	1.71	1.45	1.33
6.4 mm argon space	1.19	2.23	3.36	2.19	1.87	1.82	1.55	1.47
Quadruple Glazing, $\epsilon = 0.10$ on surfaces 2 or 3 and 4 or 5								
6.4 mm air spaces	1.25	2.28	3.40	2.23	1.91	1.86	1.59	1.52

[a]Heat transmission coefficients are based on winter conditions of −18 C outdoors and 21 C indoors with 24 km/h wind and zero solar flux. Small changes in the indoor and outdoor temperatures will not significantly affect the overall *U*-factors. Glazing layers are numbered from outdoor to indoor.

Source: Reprinted with permission from *ASHRAE Handbook, Fundamentals Volume*, 1997.

Table 5-8 Transmission Coefficients U for Wood and Steel Doors

Nominal Door Thickness in. (mm)	Description	No Storm Door	Metal Storm Door 1[a]
Wood Doors[b,c]		Btu/(hr-ft^2-F) [W/(m^2-C)]	
$1\frac{3}{8}$ (35)	Panel door with $\frac{7}{16}$ in. panels[d]	0.57 (3.24)	0.37 (2.10)
$1\frac{3}{8}$ (35)	Hollow core flush door	0.47 (2.67)	0.32 (1.82)
$1\frac{3}{8}$ (35)	Solid core flush door	0.39 (2.21)	0.28 (1.59)
$1\frac{3}{8}$ (45)	Panel door with $\frac{7}{16}$ in. panels[d]	0.54 (3.07)	0.36 (2.04)
$1\frac{3}{4}$ (45)	Hollow core flush door	0.46 (2.61)	0.32 (1.82)
$1\frac{3}{4}$ (45)	Panel door with $1\frac{1}{8}$ in. panels[d]	0.39 (2.21)	0.28 (1.59)
$1\frac{3}{4}$ (45)	Solid core flush door	0.40 (2.27)	0.26 (1.48)
$2\frac{1}{4}$ (57)	Solid core flush door	0.27 (1.53)	0.21 (1.19)
Steel Doors[c]			
$1\frac{3}{4}$ (45)	Fiberglass or mineral wool core with steel stiffeners, no thermal break[e]	0.60 (3.41)	—
$1\frac{3}{4}$ (45)	Paper honeycomb core without thermal break[e]	0.56 (3.18)	—
$1\frac{3}{4}$ (45)	Solid urethane foam core without thermal break[b]	0.40 (2.27)	—
$1\frac{3}{4}$ (45)	Solid fire-rated mineral fiberboard core without thermal break[e]	0.38 (2.16)	—
$1\frac{3}{4}$ (45)	Polystyrene core without thermal break (18-gage commercial steel)[e]	0.35 (1.99)	—
$1\frac{3}{4}$ (45)	Polyurethane core without thermal break (18-gage commercial steel)[e]	0.29 (1.65)	—
$1\frac{3}{4}$ (45)	Polyurethane core without thermal break (24-gage commercial steel)[e]	0.29 (1.65)	—
$1\frac{3}{4}$ (45)	Polyurethane core with thermal break and wood perimeter (24-gage residential steel)[e]	0.20 (1.14)	—
$1\frac{3}{4}$ (45)	Solid urethane foam core with thermal break[b]	0.20 (1.14)	0.16 (0.91)

Note: All U-factors are for exterior door with no glazing, except for the storm doors that are in addition to the main exterior door. Any glazing area in exterior doors should be included with the appropriate glass type and analyzed. Interpolation and moderate extrapolation are permitted for door thicknesses other than those specified.

[a]Values for metal storm door are for any percent glass area.

[b]Values are based on a nominal 32 × 80 in. door size with no glazing.

[c]Outside air conditions: 15 mph wind speed, 0 F air temperature; inside air conditions: natural convection, 70 F air temperature.

[d]55 percent panel area.

[e]ASTM C 236 hotbox data on a nominal 3 × 7 ft door with no glazing.

Source: ASHRAE Handbook, Fundamentals Volume. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2001.

Table 8-2 Rates of Heat Gain from Occupants of Conditioned Spaces[a]

Degree of Activity	Typical Application	Total Heat Adults, Male		Total Heat Adjusted[b]		Sensible Heat		Latent Heat	
		Btu/hr	W	Btu/hr	W	Btu/hr	W	Btu/hr	W
Seated at theater	Theater—matinee	390	114	330	97	225	66	105	31
Seated at theater	Theater—evening	390	114	350	103	245	72	105	31
Seated, very light work	Offices, hotels, apartments	450	132	400	117	245	72	155	45
Moderately active office work	Offices, hotels, apartments	475	139	450	132	250	73	200	59
Standing, light work; walking	Department store, retail store	550	162	450	132	250	73	200	59
Walking; standing	Drugstore, bank	550	162	500	146	250	73	250	73
Sedentary work[c]	Restaurant	490	144	550	162	275	81	275	81
Light bench work	Factory	800	235	750	220	275	81	475	139
Moderate dancing	Dance hall	900	264	850	249	305	89	545	160
Walking 3 mph; light machine work	Factory	1000	293	1000	293	375	110	625	183
Bowling[d]	Bowling alley	1500	440	1450	425	580	170	870	255
Heavy work	Factory	1500	440	1450	425	580	170	870	255
Heavy machine work; lifting	Factory	1600	469	1600	469	635	186	965	283
Athletics	Gymnasium	2000	586	1800	528	710	208	1090	320

[a]Tabulated values are based on 75 F room dry bulb temperature. For 80 F room dry bulb, the total heat remains the same, but the sensible heat values should be decreased by approximately 20 percent, and the latent heat values increased accordingly.

[b]*Adjusted heat gain* is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85 percent of that for an adult male, and that the gain from a child is 75 percent of that for an adult male.

[c]Adjusted total gain for *sedentary work, restaurant*, includes 60 Btu/hr for food per individual (30 Btu/hr sensible and 30 Btu/hr latent).

[d]For *bowling*, figure one person per alley actually bowling, and all others sitting (400 Btu/hr) or standing and walking slowly (550 Btu/hr).

Source: Reprinted by permission from *ASHRAE Cooling and Heating Load Calculation Manual*, 2nd ed., 1992.